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G1N NAAJCR N7N
F2L LP L211 L218 L301 L302 L501

(56) Documents Cited
GB 2325036 A GB 2317933 A
GB 2313885 A GB 2309761 A
GB 2297596 A US 5832777 A

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UK CL (Edition T) F2L LK LP, G1N NAAJCR NAAJR
INT CL^7 F16D 27/00 48/06, G01M 13/02
ONLINE: WPI,EPODOC,JAPIO

(54) Abstract Title

Calibrating a balance position where a compensation spring is used to balance a resilient load operated by an actuator

(57) A method for calibrating the force balance position at which a compensating spring 60 of an electric motor driven actuator 30 balances the resilient load of the diaphragm spring 90 of a clutch (16, Fig 1). An alternating positional signal is applied to the electric motor 40 which corresponds to the positions of a push rod 48 of the actuator 30 which span the balance position, whilst the actual position of the push rod 48 of the actuator 30 is measured via a positional sensor 68.

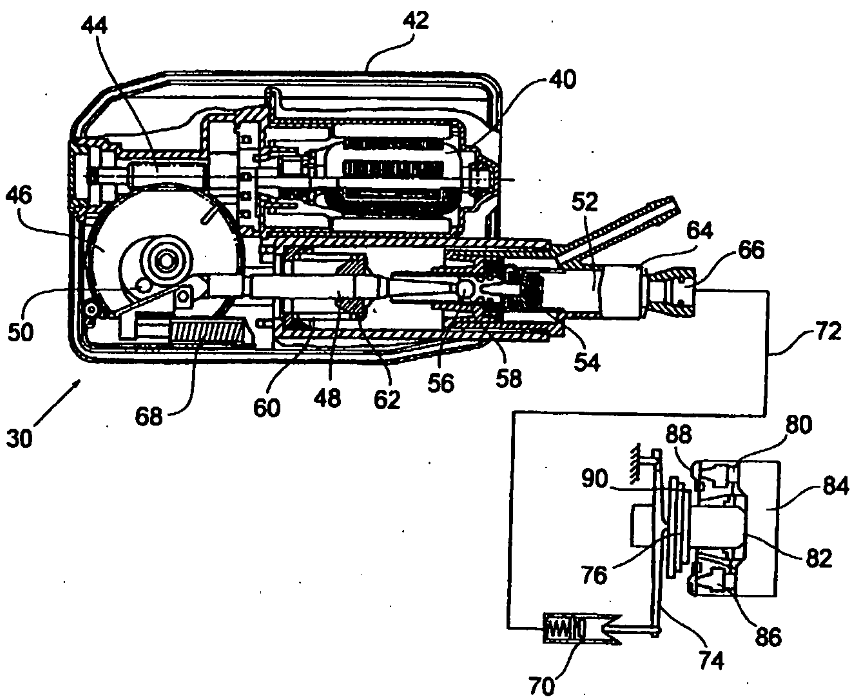

Fig 2.

GB 2 374 421 A

The print reflects an assignment of the application under the provisions of Section 30 of the Patents Act 1977.

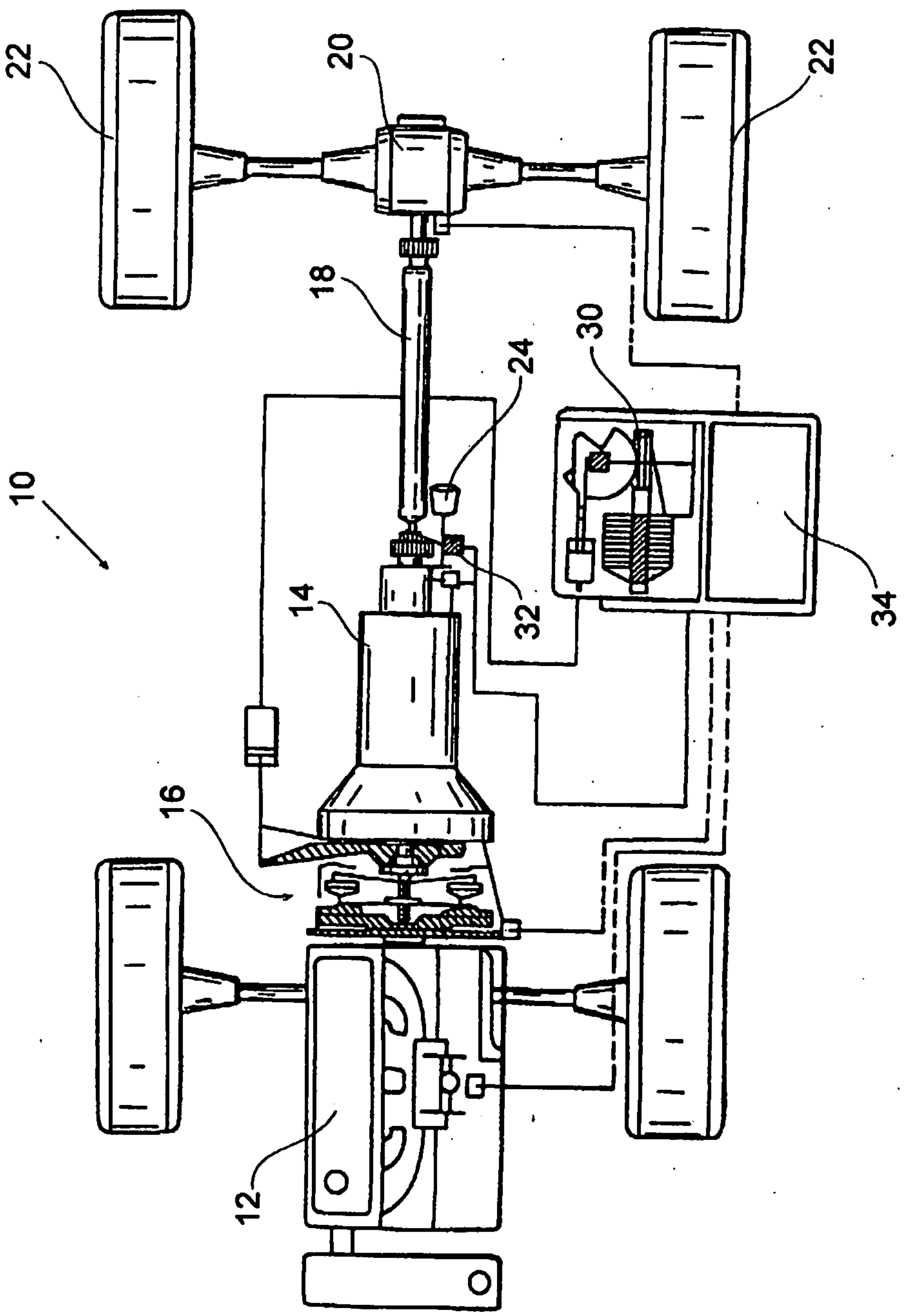

Fig 1.

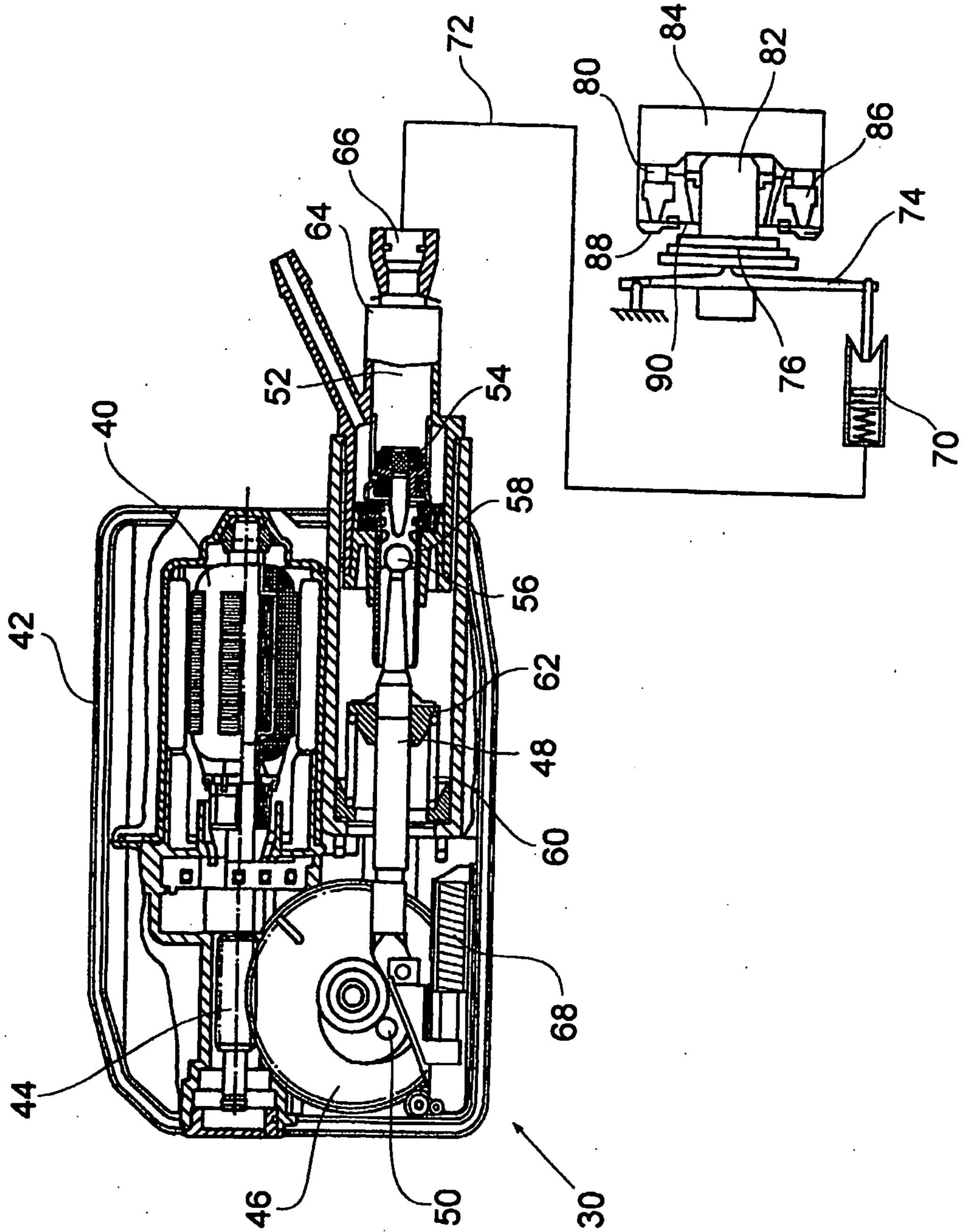

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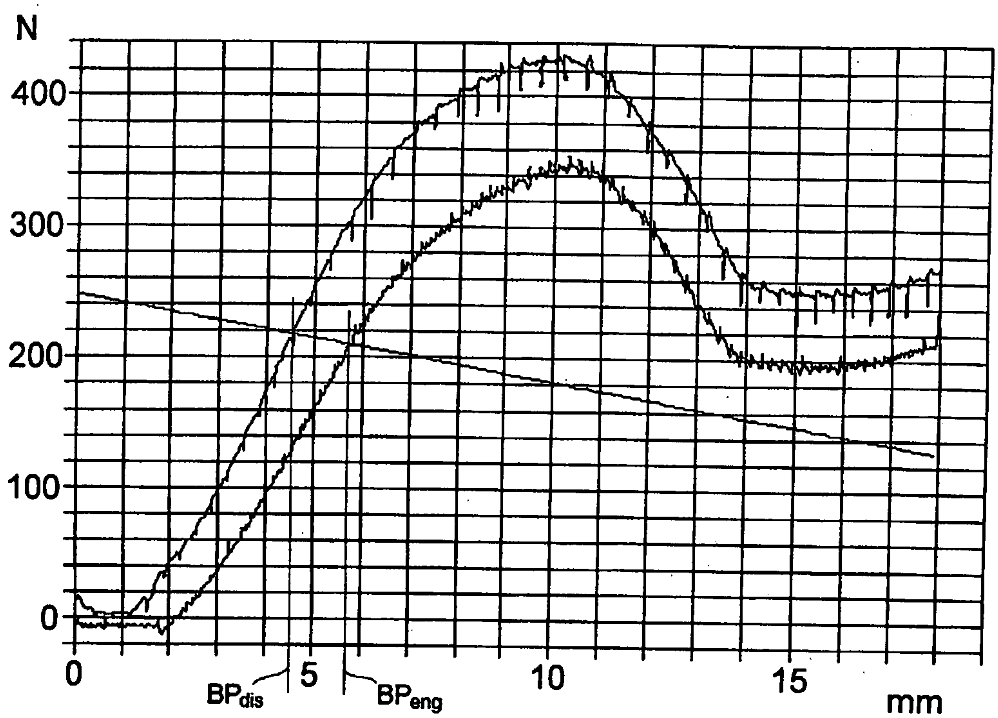
Fig 2.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mm
BPdis
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Fig 3.

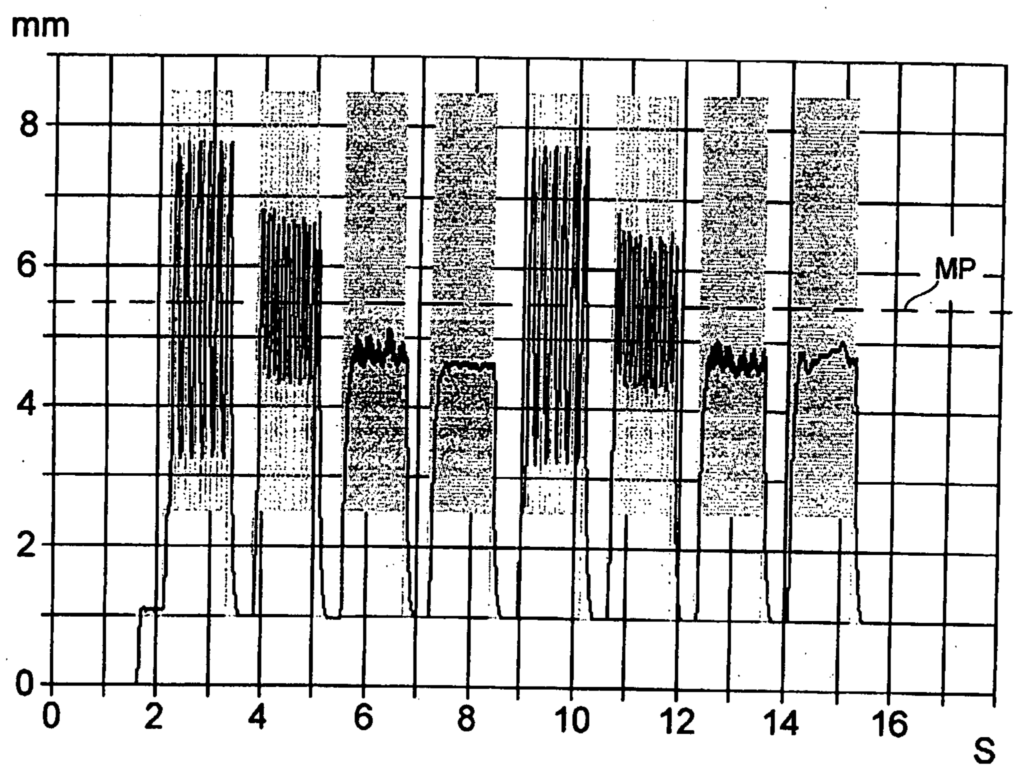
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Fig 4.

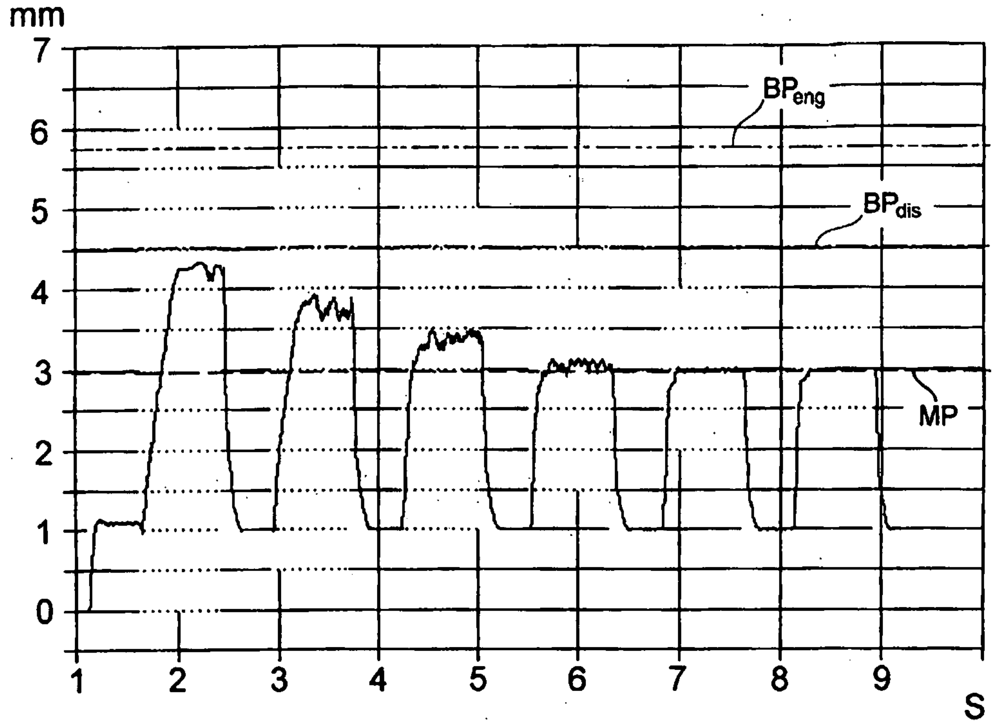
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Fig 5

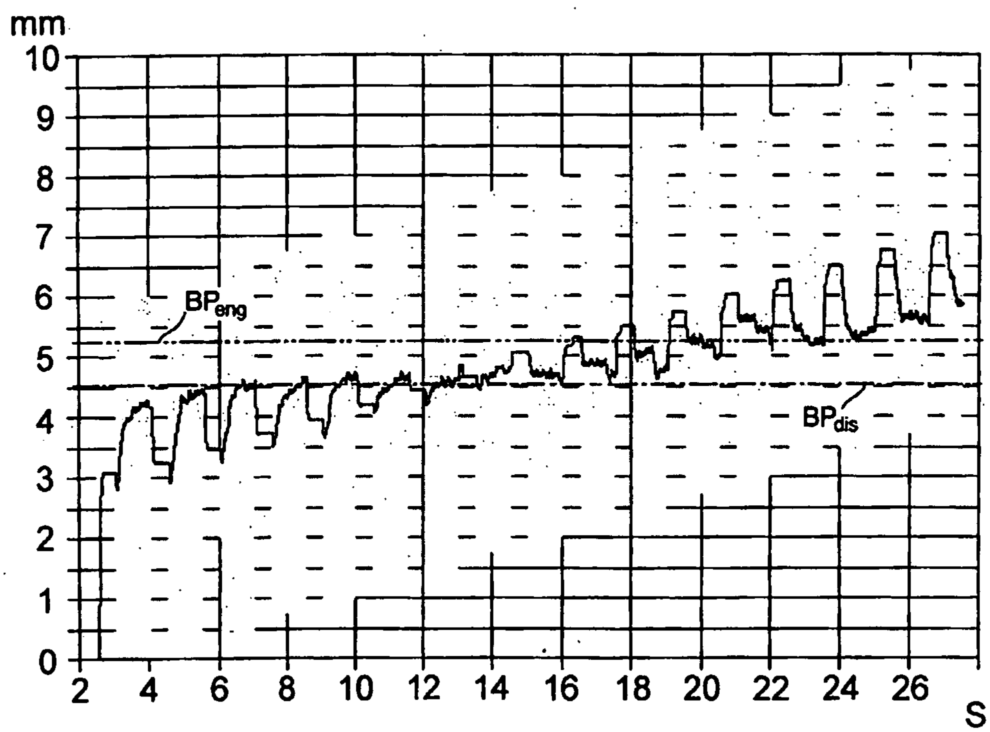
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Fig 6

ELECTRICALLY OPERATED ACTUATOR

The present invention relates to electrically operated actuators and in particular to an electrically operated actuator for a motor vehicle clutch or gear selector mechanism.

Electrically operated actuators in accordance with the present invention, for example as disclosed in GB2325036, GB2313885 or GB2309761 the disclosure of which explicit reference is made and whose content is expressly incorporated in the disclosure content of the present application, comprise an electric motor for controlling actuation of a hydraulic master cylinder and, via a slave cylinder, a vehicle clutch or gear ratio selector mechanism. In such actuators, the electric motor may be connected via a suitable gearbox, and via, for example a worm and worm gear mechanism, to a pushrod which is connected to the worm gear by means of a crank, so that the rotation of the worm gear is transmitted into linear motion of the pushrod. The free end of the pushrod is attached to a piston which is slidably sealed in a master cylinder. The motor, gear mechanism and master cylinder are preferably all provided in a common housing.

The master cylinder of the electric actuator described above will typically be connected to a clutch slave cylinder so that application of pressure to the clutch slave cylinder will operate a clutch release fork which acts on a clutch release bearing, to apply a load to release the clutch. The release bearing would typically act on a diaphragm spring which normally maintains the plates of the clutch in frictional engagement, depression of the diaphragm spring causing the clutch plates to move apart, so that the clutch is disengaged. The load generated by the electric motor must consequently be capable of depressing the diaphragm spring to an extent sufficient to disengage the clutch. The load generated by the electric motor to disengage the clutch may typically be of the order of 450 N.

In order to reduce the size of the electric motor required for such actuators, it has been proposed to include a compensating spring in the electric actuator, said compensating spring opposing the load applied by the diaphragm spring. The compensating spring, may, for example, be arranged to be fully compressed when the clutch is fully engaged, exerting a load of say 250 N, which acts on the electric actuator in the direction of movement of the actuator to disengage the clutch. Upon disengagement of the clutch, the initial load to depress the diaphragm spring will then be supplied by the compensating spring and while this load will have reduced by the time the compensating spring and diaphragm spring are in balance, the electric motor is only required to provide a load of the order of 200 N to fully disengage the clutch. The electric motor may thus be down-rated from 450 to 500 N which would be required without the compensating spring to 250 to 300 N by use of the compensating spring.

With electric actuators of the type disclosed high static friction in the worm gear will provide a self holding effect. If however the internal static friction of the actuator is insufficient, which may be desirable in order to optimise efficiency of the actuator, it is possible when the actuator is at rest, that the load applied by the diaphragm spring when the clutch is disengaged will force the actuator back, or the load applied by the compensating spring when the clutch is engaged will force the actuator forward. If the resulting difference between the actual and requested position of the actuator is greater than a predetermined tolerance, then the controller will reactivate the actuator.

In order to compensate the force applied by the diaphragm spring which may allow the clutch to re-engage unintentionally, it has been proposed in German Patent Application No DE10062456.1 the disclosure of which explicit reference is made and whose content is expressly incorporated in

the disclosure content of the present application, when the actuator is at rest, to apply a voltage of typically 7 per cent of the maximum of PWM voltage to the electric motor in a direction supporting the compensating spring and opposing the diaphragm spring. This voltage applies a load to the actuator which prevents backward movement of the actuator under the load applied by the diaphragm spring. However, in order to avoid forward movement of the actuator when the clutch is engaged, the 7 per cent voltage is only applied when the load applied by the compensating spring is lower than that applied by the diaphragm spring.

It is consequently a requirement of this system, to know the balance position between the compensating spring and balance spring. For clutches with self adjusters which adjust the position of the pressure plate to accommodate wear of the friction faces, the balance position remains substantially fixed throughout the life of the clutch. With such clutches, the balance position may be precalibrated. However, where the clutch is not fitted with self adjusters, the balance position will vary significantly as the components of the clutch wear. With such clutches, the 7 per cent voltage strategy described above cannot used and a far more complex adaptive continuous current feed strategy must be implemented.

The present invention provides a method of calibrating the force balance position between the clutch and compensating spring, once the system has been installed in a vehicle, so that the balance position may be calibrated in an end-of-line procedure and recalibrated at regular intervals, for example routine vehicle service intervals, during the life of the vehicle. This enables the 7 per cent PWM voltage strategy to be used on vehicles with clutches which do not have self adjusters.

According to one aspect of the present invention, a method of calibrating the balance position at which a compensating spring of an electric motor

driven actuator balances a resilient load in the mechanism which is operated by the actuator, comprises applying a large amplitude, high frequency alternating positional signal to energise the electric motor, said positional signal corresponding to positions of the actuator which span the balance position, and measuring the actual position of the actuator by means of a positional sensor associated with the actuator.

Because the force characteristic of the compensating spring and resilient load applied by the mechanism, upon initial energisation in one direction, when for example movement of the actuator is assisted by the compensating spring, the actuator will move rapidly to the balance position, the rate of movement will then be significantly reduced as the motor opposes the load applied by the resilient load of the mechanism. Similarly, upon reverse energisation of the motor the actuator will move quickly back to the balance position and then slowly beyond the balance position, as the compensating spring is compressed. Consequently, the actual position of the actuator centres on the balance position. The higher the frequency of the alternating signal, the shorter the distance moved beyond the balance point in each direction and consequently the better the resolution of the balance position. According to a preferred embodiment the frequency of the alternating positional signal is 25 Hz or greater, more preferably a positional signal of about 50 Hz is used.

The closer the mid-point of the positional signal is to the balance position, the more accurate the balance position determined. An iterative technique may consequently be used, carrying out successive determinations of the balance position, positioning the mid-point of the alternating positional signal on the previously determined balance position, until the balance position coincides with the mid-point of the alternating positional signal. Initially, the mid-point of the alternating positional signal may be arranged to coincide with a theoretical balance position calculated from the design characteristics of the actuator/mechanism or at least the

last known balance position, where the actuator/mechanism are being recalibrated.

An embodiment of the invention is now described, by way of example only, with reference to the following drawings, in which:-

Figure 1 illustrates diagrammatically a vehicle with an electric motor driven clutch actuator;

Figure 2 illustrates in greater detail the electric motor driven clutch actuator/clutch of the vehicle illustrated in Figure 1;

Figure 3 shows typical plots for load against actuator travel for the clutch diaphragm spring and actuator compensating spring illustrated in Figure 2;

Figure 4 shows a plot of the actual actuator output position when alternating high amplitude positional signals are applied to the actuator, at varying frequencies;

Figure 5 is a plot of actual actuator output position when a high frequency alternating positional signal is applied to the actuator, at varying amplitudes; and

Figure 6 is a plot of actual actuator output position when the mid-point of a high amplitude, high frequency positional signal is applied to the actuator, the mid-point of the alternating positional signal being varied relative to the balance position of the actuator/clutch connection.

As illustrated in Fig. 1, a vehicle 10 has an internal combustion engine 12, which is connected to a gearbox 14 via a clutch 16. The gearbox 14 is connected by a driveshaft 18 and rear axle 20 to drive the rear wheels 22 of the vehicle 10.

A gear selector lever 24 is connected mechanically to the gearbox 14 in conventional manner, for manual selection of the gear ratio. Engagement and disengagement of the clutch 16 is controlled by an electric motor driven clutch actuator 30, a sensor 32 on the gear selector lever 24 providing a control signal to a control unit 34, which will activate the clutch actuator 30 to disengage and re-engage the clutch 16 as appropriate, when a change of gear ratio is initiated by movement of the gear selector lever 24.

As illustrated in Fig. 2, the clutch actuator 30 comprises a direct current electric motor 40 mounted to the housing 42. The motor 40 is connected either directly or via a fixed ratio gear mechanism, a worm 44 and a worm gear wheel 46 to a push rod 48, the pushrod 48 being connected to a crank 50 associated with worm gear wheel 46 so that rotation of the worm gear wheel 46 is translated into linear motion of the pushrod 48. Instead of the worm gearing 44,46, other forms of connection may be used to transmit the rotary motion of the electric motor 40 to the pushrod 48, for example planetary gearing, spur wheel gearing, cone pulley gearing or gearing with threaded spindle may alternatively be used.

The push rod 48 is connected at its free end to a piston 54 of a hydraulic master cylinder 52, the hydraulic master cylinder 52 being formed integrally of the electric motor housing 42. The push rod 48 is connected to the piston 54 by means of a ball formation 56, which is a snap-fit within a part spherical recess 58 formed axially of the piston 54. A helical compression spring 60 acts between the housing 42 and a collet 62 mounted on the pushrod 48, to urge the push rod 48 towards the closed end 64 of the master cylinder 52. A port 66 opens to the master cylinder 52 at the end 64 thereof.

A position sensor 68 in the form of a linear potentiometer is mounted for movement with the push rod 48 to provide a signal indicative of the position of push rod 48

The port 66 of the master cylinder 52 is connected to a clutch slave cylinder 70 by a hydraulic line 72. The slave cylinder 70 acts on a clutch release fork 74 which operates on release bearing 76 to control engagement and disengagement of clutch 16 in conventional manner.

The clutch 16 comprises of friction plate 80 which is diagonally connected to the input shaft 82 to the gearbox 14. The friction plate 80 is mounted coaxially and between a flywheel 84 which is drivingly connected to the engine and a pressure plate 86 which is connected via a clutch housing 88 to the flywheel 84 for rotation therewith, the pressure plate 86 being movable axially with respect to the flywheel 84. The pressure plate 86 is urged towards the flywheel 84, so that the friction plate 80 will be clamped therebetween in order to transmit torque between the engine 12 and gearbox 14, by means of a diaphragm spring 90. The clutch 16 is released by application of a load towards the flywheel 84 to the internal diameter of the diaphragm spring 90, by means of the release fork 74 and release bearing 76.

Instead of having a hydraulic link between the clutch actuator 30 and release fork 74, a clutch actuator 30 may be connected thereto by a pneumatic link, or mechanically, for example the pushrod 48 may act directly on the release fork 74 or may be connected thereto by means of a mechanical linkage or cable.

When clutch 16 is fully engaged the clutch actuator 30 will be in the position illustrated in Fig. 2, with the push rod 48 shifted hard over to the left, so that the piston 54 of the master cylinder 52 is at its limit of

movement away from end 64 thereof and the spring 60 is fully compressed.

When the electric motor 40 is energised to disengage the clutch 16, the pushrod 48 is moved to the right, so that piston 54 moves towards end 64 of master cylinder 52. Fluid is thereby displaced from the master cylinder 52 to the slave cylinder 70. The slave cylinder 70 will thereby apply a load to the release fork 74 which moves the release bearing 76 towards the flywheel 84 and applies the load to the inner periphery of the diaphragm spring 90 to reduce the load applied thereby to the pressure plate 86, reducing the clamping load on the friction plate 80.
As illustrated in Fig. 3, initially the load applied by the fully compensated spring 60 will be in excess of the reaction load of the diaphragm spring 90 and consequently push rod 48 and piston 54 will move under the load applied by the spring 60. The electric motor 40 will consequently be under very little load acting only to permit movement of the pushrod 48 under the action of spring 60.

At the balance position when the load applied by the spring 60 balances the reaction load of the diaphragm spring 90, the load required for further disengagement of the clutch 16 will then be provided by the electric motor 42. As indicated in Fig. 3, the load applied to the diaphragm spring 90 in order to fully disengage the clutch 16 will typically be of the order of 430N. The compensating spring 60 is rated to provide a load of the order of 250N when the clutch is fully engaged and the reaction load of the diaphragm spring 90 is substantially zero, the load applied by the compensating spring 60 falling to about 210N at the balance position. Electric motor 40 must consequently be capable of applying a load sufficient to depress the diaphragm spring 90 from the balance position to the fully disengaged position of the clutch, that is from 210N to 430N and to fully compress the spring 60 from the balance position to the fully engaged position of the clutch. An electric motor 40 rated to provide a

load of 220 to 250N will consequently be suitable rather than requiring a motor 40 capable of producing loads in excess of 430N. As illustrated in Fig. 3, due to the hysteresis of the diaphragm spring 90, the balance position upon disengagement of the clutch 16 (BP_{dsi}) is different from the balance position upon engagement of the clutch 16 (BP_{eng}), being 4.5mm and 5.7mm respectively.

With electric actuators of the type disclosed above, unless there is significant friction in the mechanism, the loads applied by the compensation spring 60 when the clutch is fully engaged or the diaphragm spring 90 when the clutch is fully disengaged, will cause the electric motor 40 when de-energised to wind back, thus permitting the actuator 30 to move from the required position. If, during a gear change, the actual position of the actuator 30 varies from the required position by more than a predetermined amount, the electric motor 40 will be re-energised to move the actuator 30 back to the desired position. In order to avoid this when the actuator 30 is at rest it has been proposed to apply a current to the electric motor 40 sufficient to hold the motor 40 in position but not sufficient to cause movement of the actuator 30. Typically, a voltage of 7% of the normal PWM voltage is applied to the electric motor 40 for this purpose. This 7% PWM strategy is however only used when the actuator 30 is at rest during a gear change, with the actuator 30 between the balance position and the fully disengaged position of the clutch 16. It is consequently a requirement when using this strategy to know accurately the balance position of the actuator/clutch assembly.

As the balance position of the actuator/clutch assembly varies with wear of the friction faces of the clutch 16, there is a requirement to calibrate the actuator/clutch assembly, throughout the life of the vehicle.

Fig. 4 illustrates the effect of applying a position signal alternating at frequencies of 5Hz, 10Hz, 25Hz and 50Hz to the electric motor 40 of the actuator 30. The positional signal is of 6mm amplitude, the mid-point (MP) of the oscillation being around the predicted balance point of the actuator 30/clutch 16 assembly. As the positional signal is applied from the fully engaged position of the clutch 16, the electric motor 30 assisted by the compensating spring 60 will move the actuator 30 rapidly to the balance position. The actuator 30 will then move more slowly as the motor 40 itself acts to overcome the reaction force of the diaphragm spring 90. As a consequence, even at a frequency of 5Hz, the actuator will not reach the position required by the positional signal, before the positional signal is reversed. Upon reversal of the positional signal, the electric motor 40 assisted by diaphragm spring 90 rapidly returns the actuator 30 to the balance position and then moves more slowly as the motor 40 compresses spring 60. The higher the frequency of the positional signal the less the actuator overshoots the balance position and at frequencies of 25 Hz and 50Hz, the actuator 30 settles down at the balance position, as illustrated by Fig. 4.

As illustrated in Fig. 5, a positional signal alternating at a frequency of 50Hz and of varying amplitude is applied to the actuator 30. As shown in Fig. 5, at smaller amplitudes the actuator 30 will settle down at the mid-point of the alternating positional signal, the larger the amplitude the closer the actuator 30 settles down at a position corresponding to the balance position of the actuator 30.

Finally, as illustrated in Fig. 5, in which a positional signal alternating at a frequency of 50Hz and having an amplitude of 6mm, is applied to the actuator 30, the mid-point of the oscillation of the positional signal being varied, it is shown that the accuracy of the balance position determination improves as the mid-point of the positional signal approaches the balance position.

According to one embodiment of the invention, to calibrate the balance position of an actuator 30, a positional signal alternating at a frequency of 50Hz and of 6mm amplitude is applied to the actuator 30 and the actual position of the actuator 30, as indicated by the positional sensor 68, is determined. Initially, the mid-point of the alternating positional signal is set to coincide with a calculated or previously determined balance position.

The calibration procedure is then repeated, the mid-point of the positional signal being repositioned to coincide with the balance position determined in the previous calibration procedure, until the balance position determination coincides with the mid-point of the positional signal.

Various modifications may be made without departing from the invention. For example, while a positional signal alternating at 50Hz is used in the preferred embodiment described above, positional signals alternating at 25Hz or more may be used. Furthermore, while an amplitude of 6mm is used in the above embodiment, it will be appreciated that the actual amplitude used will depend upon the amount of movement of the actuator from the fully engaged to the fully disengaged position of the clutch and the location of the balance position relative to these extremes of movement.

While the invention has been described with reference to a clutch actuator, it is equally applicable to other electronic motor actuators which include a compensating spring and are used to control movement of a mechanism which produces a resilient reactional force, for example actuators used in gear selector mechanisms. Actuators according to the present invention may also be used in fully or semi-automated transmission systems.

The patent claims submitted with the application are proposed formulations without prejudice to the achievement of further patent protection. The applicant reserves the right to submit claims for further combinations of characteristics, previously only disclosed in the description and/or drawings.

References back used in sub-claims refer to the further development of the subject of the main claim by the characteristics of the respective sub-claim; they are not to be understood as a waiver with regard to achieving independent item protection for the combination of characteristics in the related sub-claims.

Since the subject of the sub-claims can form separate and independent inventions with reference to the prior art on the priority date, the applicant reserves the right to make them the subject of independent claims or of division declarations. Furthermore, they may also contain independent inventions which demonstrate a design which is independent of one of the objects of the preceding sub-claims.

The embodiments are not to be considered a restriction of the invention. Rather, a wide range of amendments and modifications is possible within the scope of the current disclosure, especially those variations, elements and combinations and/or materials which, for example, the expert can learn by combining individual ones together with those in the general description and embodiments in addition to characteristics and/or elements or process stages described in the claims and contained in the drawings with the aim of solving a task thus leading to a new object or new process stages or sequences of process stages via combinable characteristics, even where they concern manufacturing, testing and work processes.

CLAIMS

1. A method of calibrating the balance position at which a compensating spring of an electric motor driven actuator balances a resilient load in the mechanism which is operated by the actuator, comprises applying a large amplitude, high frequency alternating positional signal to energise the electric motor, said positional signal corresponding to positions of the actuator which span the balance position, and measuring the actual position of the actuator by means of a positional sensor associated with the actuator.

2. A method according to claim 1 in which the alternating positional signal alternates at a frequency of 25Hz or greater.

3. A method according to claim 1 or 2 in which the alternating positional signal alternates at a frequency of the order of 50Hz.

4. A method according to any one of claims 1 to 3 in which the alternating positional signal has an amplitude of the order of 6mm.

5. A method according to any one of claims 1 to 4 in which the mid-point of the alternating positional signal is positioned to coincide with an estimated balance position.

6. A method according to any one of the preceding claims in which successive calibrations are carried out, at each calibration following the initial calibration, the mid-point of the alternating positional signal is repositioned to coincide with the balance position determined by the previous calibration, until the balance position determined coincides with the mid-point of the alternating positional signal.

Application No:	GB 0109222.0	**Examiner:**	Richard Kerslake
Claims searched:	1-6	**Date of search:**	11 January 2002

14

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.T): G1N (NAAJCR, NAAJR); F2L (LP, LK)

Int Cl (Ed.7): G01M 13/02; F16D 27/00, 48/06

Other: Online: WPI, EPODOC, JAPIO

Documents considered to be relevant:

Category	Identity of document and relevant passage		Relevant to claims
A	GB 2325036 A	(LUK GETRIEBE-SYSTEME GmbH)	1
Y	GB 2317933 A	(LUK GETRIEBE-SYSTEME GmbH)	1
Y	GB 2313885 A	(LUK GETRIEBE-SYSTEME GmbH)	1
A	GB 2309761 A	(LUK GETRIEBE-SYSTEME GmbH)	1
A	GB 2297596 A	(FICHTEL & SACHS AG)	1
X	US 5832777	(WEILANT)	1

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
Y	Document indicating lack of inventive step if combined with one or more other documents of same category.	P	Document published on or after the declared priority date but before the filing date of this invention.
&	Member of the same patent family	E	Patent document published on or after, but with priority date earlier than, the filing date of this application.